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Update on Results of SPRE Testing at NASA Lewis

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ABSTRACT

The Space Power Research Engine (SPRE), a free-piston Stirling engine with a linear alternator, is being tested at NASA Lewis Research Center as part of the Civilian Space Technology Initiative (CSTI) as a candidate for high capacity space power. This paper presents results from recent SPRE tests designed to investigate the effects of variation in the displacer seal clearance and piston centering port area on engine performance and dynamics. The impact of these variations on PV power and efficiency are presented. Comparisons of the displacer seal clearance test results with HFAST code predictions show good agreement for PV power, but show poor agreement for PV efficiency. Correlations are presented relating the piston mid-stroke position to the dynamic ΔP across the piston and the centering port area. Test results indicate that a modest improvement in PV power and efficiency may be realized with a reduction in piston centering port area.

INTRODUCTION

In support of the NASA Civil Space Technology Initiative High Capacity Power Program, NASA Lewis Research Center is testing the free-piston Stirling engine in combination with a linear alternator as a candidate conversion system for advanced space electrical power generation. Much of this testing is being conducted using the Space Power Research Engine (SPRE), built under contract to NASA by Mechanical Technology, Inc. of Latham, NY. This engine was originally one half of the Space Power Demonstrator Engine, which was split and modified into two separate engines. The SPRE engine and some of the NASA baseline engine testing are described in some detail in references [1-6]. Much of the work in the past year has been focused on producing experimental data for validating the HFAST engine performance code, being developed by MTI as part of the NASA contract, as well as for developing other useful design tools. Two areas of particular interest have been: 1) the sensitivity of the engine performance to the displacer seal clearance and 2) the effects of varying the piston centering port area. This paper reports results of testing with the SPRE to investigate these areas. Experimental results from displacer seal clearance tests for PV power and efficiency are compared to HFAST code predictions. The results of testing with several different piston centering port areas are

used to develop simple correlations relating the piston mid-stroke position to measured parameters and to the centering port area.

DISPLACER SEAL CLEARANCE TESTS

Objectives:

The objectives of the displacer seal clearance tests were to:

1. determine the sensitivity of the SPRE PV power and efficiency to the displacer seal radial clearance,

2. obtain a detailed parametric data base for comprehensive comparison with results of the HFAST code.

Test Description:

The original plan was to test the SPRE with the displacer seal clearance reduced to 1 mil to determine if the PV power and efficiency would increase. However, a measurement check of the displacer cylinder runout relative to the displacer bearing indicated that the cylinder bore was wavy and had a total indicated runout on the order of 2 mils. Therefore, the clearance could not be reduced significantly below the baseline 2 mil clearance without the displacer rubbing against the cylinder wall. Rework of the cylinder to improve roundness and concentricity was not considered practical. The approach selected was to cut a groove in the displacer cylinder and install a compliant Rulon seal ring, at the mid-point of the clearance seal, to minimize the leakage past the displacer. Figure 1 is a photo of the modification made to the displacer cylinder. Figure 2 is a photo of the seal ring and the wavy backing spring used to hold the seal against the displacer.

After the initial test with the displacer seal ring, it soon became apparent that the seal ring friction effected the displacer dynamics, although considerable effort had been made to minimize the drag of the seal on the displacer. The seal ring friction made it nearly impossible to separate seal leakage effects from the effects caused by friction attenuation of the displacer motion. So, the seal ring was discarded.

At this point it was decided to investigate the effect of in-

creasing the displacer seal clearance. A series of tests were performed, starting with the baseline seal radial clearance, with an incremental increase in the clearance until it produced a significant measurable effect on PV power and/or PV efficiency. Therefore, tests were performed with the baseline clearance of 2 mil (0.002 inch),and 3, 4 and 5 mils nominal displacer seal radial clearance. The clearance was changed by grinding the very hard chrome oxide coated outside diameter of the displacer base, with a diamond wheel, to achieve the desired radial clearance with the cylinder. Figure 3, a photo of the displacer, shows the area on the base which was ground to adjust the seal clearance. Care was taken to ensure concentricity of the ground surface to the displacer bearings. Typical measured total runout of the ground surface, with the displacer assembled and rotated in its bearings, was less than 0.2 mils.

Results:

Figure 4 shows the effect of the displacer seal clearance on the measured PV power with the engine operating at 2.0 temperature ratio and 15 MPa. Experimental data is shown for both 8 and 10 mm piston amplitudes. The HFAST code predicted values shown are for 9 mm piston amplitude based on a displacer/piston amplitude ratio of .933 and 81° displacer phase angle relative to the piston motion. The actual measured values for displacer/piston amplitude ratios, for the 8 and 10 mm piston amplitude points, ranged from ~1.03 to 1.15 and values for displacer phase angles ranged from 79.36° to 80.2°. The PV power tended to slowly diminish as the displacer seal clearance increased, and this change in PV power agreed reasonably well with the predictions. The measured power at the design condition (2.0 temperature ratio, 15 MPa mean pressure, and 10 mm piston amplitude), with 5 mil clearance was ~505 W lower than the power observed with 2 mil clearance. Using linear interpolation to adjust for a 10 mm piston amplitude, the code predicted ~456 W lower power.

The measured and predicted effects of increased displacer seal clearance on the PV efficiency is shown in Figure 5. There was no apparent change in measured PV efficiency with increased displacer seal clearance, which disputes the code prediction of a very significant efficiency drop. It should be noted that, in order to have the HFAST code converge with clearances larger than 3 mils, some of the appendix gap calculations had to be suppressed. This, in combination with effects that would result from the difference between the measured displacer motions and those upon which the predicted values were based, may account for the apparent disparity.

PISTON CENTERING PORT TESTS

Objectives:

The objectives of the piston centering port tests were to:

1. determine the piston mid-stroke position sensitivity to the centering port area over a wide range of SPRE operating conditions, i.e. temperature ratio, mean pressure, and piston amplitude,

2. develop a correlation model for the piston mid-stroke position based on the experimental data,

3. determine the effects of the piston centering port area on PV power and efficiency.

Test description:

The SPRE piston has two sets of mid-stroke centering ports as illustrated, schematically, in Figure 6. One set of centering ports, located at the compression space end, connects the compression space to the gas bearing drain plenum. The second set of ports, which is located at the gas spring end, connects the piston gas spring to the gas bearing drain plenum. For these tests, only the effects of the centering ports on the compression space end of the piston were investigated. These ports, shown in the photo in Figure 7, are L-shaped passages consisting of equally spaced holes, located near the outer diameter of the piston, which are drilled axially from the compression space end of the piston and intersect with radially drilled holes through the outer diameter. These ports can easily be blocked off by installing a set screw in the compression space end of the passage, which is threaded.

The centering port tests were run concurrently with some of the displacer seal clearance tests, and were performed after the displacer cylinder was modified by adding the groove for the seal ring. Therefore, the clearance seal data with 2 mil displacer clearance, both with and without the displacer seal installed, were used for baseline references with 6 piston centering ports open. Tests also were performed with 4 centering ports, without the displacer seal, and with 3 and 2 centering ports open, with the displacer seal installed. For all tests, the plugs were arranged so that the open ports were equally spaced for flow symmetry. These centering port tests were performed with the engine operating over a wide range of conditions, where the temperature ratio varied from 1.6 to 2.0, the mean pressure varied from 5 to 15 MPa, and the piston amplitude varied from 6 to 10 mm.

Results:

Figure 8 shows the measured response of the piston mid-stroke position to variation of piston amplitude for the SPRE operating at a temperature ratio of 2.0 for 5, 10 and 15 MPa with 6 and with 2 centering ports. Figure 9 is a similar plot showing the piston mid-stroke position versus piston amplitude for the tests with 6 and 2 centering ports, but at 15 MPa mean pressure and temperature ratios ranging from 1.7 to 2.0. From these plots, it is apparent that the mid-stroke position is strongly influenced by the number of centering ports, the mean pressure, and also, to a lesser degree, by the temperature ratio. In Figure 10, the piston mid-stroke positions for all the data points with both 6 and 2 centering ports are plotted against the dynamic ΔP across the piston (the part of the vector sum of the harmonic components of the compression space pressure and piston gas spring pressure which is in phase with the piston motion). The data for 6 and 2 centering ports correlates fairly well using least-squares linear regression. The same

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correlation method also was used for the data taken with 4 and 3 centering ports. The first order curve fits for each set of data are shown in Figure 11. The slope of the lines is very sensitive to the number of open ports. The correlation coefficients (slopes) determined for each set of data are plotted versus the number of ports in Figure 12. The data points fall in nearly a straight line indicating that the slopes for each set of centering port data are proportional to the number of centering ports raised to approximately the -0.775 power. Although this correlation approach is crude, it provides a way to estimate the effective stiffness or spring rate of the centering ports, which is proportional to the reciprocal of the correlation coefficient. The effect of the centering port stiffness was somewhat apparent while performing these tests. As the number of ports was reduced, the piston mid-stroke position had a greater tendency to wander, especially at the higher power points.

The effect of the number of centering ports on PV power and efficiency was obscured by having the displacer seal installed for part of the tests, and the raw data results tend to be scattered and conflicting. Table 1 compares the measured PV power and efficiency at the design operating condition for 6 and 4 centering ports without the displacer seal, and for 6, 3, and 2 centering ports with the displacer seal. Installing the displacer seal produced a 318 W decrease in PV power and efficiency increased by 0.139 points for the baseline with 6 ports. The HFAST code predictions, based on the actual measured operating conditions, but not adjusted for the number of centering ports also are shown. The measured values can be compared after normalizing, by subtracting out the predicted values (Δ PV power and efficiency). Reducing the ports from 6 to 4, without the displacer seal ring installed, decreased the PV power 100 W and PV efficiency increased 0.211 percentage points. With the displacer seal ring installed, reducing the number of ports from 6 to 3 produced a 50 W decrease in PV power and a 0.148 point decrease in PV efficiency, while the PV power increased ~500 W and PV efficiency increased by 0.726 percentage points by reducing the ports from 6 to 2.

CONCLUDING REMARKS

The results of the SPRE displacer seal clearance tests suggest,

that since the Stirling cycle power is only mildly degraded by increasing the displacer seal clearance, and the cycle efficiency appears to be insensitive, in the range which was tested, the design clearance and tolerances for the displacer could be increased. This would reduce fabrication costs and relieve some of the concern about maintaining close running clearances for long operating periods.

Although the test results are conflicting and strongly influenced by the presence of the displacer seal ring, it appears that a modest increase on both PV power and efficiency may be achieved by reducing the piston centering port area. It should be pointed out that reduction in the number of piston centering ports will be accompanied by a decrease in the effective centering port stiffness, which can impact the dynamic response and stability of the system.

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[2] J.G. Slaby, "Overview of the 1985 NASA Lewis Research Center SP-100 Free-Piston Stirling Engine Activities, "NASA TM-87028, DOE/NASA/1005-5, 1985.

[3] J.G.Slaby, "Overview of Free-Piston Stirling Technology at the NASA Lewis Research Center," NASA TM-87156, DOE/NASA/1005-7, 1985.

[4] J.E Cairelli, "SPRE I Free-Piston Stirling Engine Testing at NASA Lewis Research Center," NASA TM-100241, 1988.

[5] J.E. Cairelli, et al., "Results From Baseline Tests of the SPRE I and comparison With Code Predictions," NASA TM-102044, 1989.

[6] J.E. Cairelli, et al., "Update on Results of SPRE Testing at NASA," Proceedings of the 25th Intersociety Energy Conversion Conference, August, 1990, pp 237-244.

TABLE 1 - EFFECTS OF THE NUMBER OF PISTON CENTERING PORTS AT THE DESIGN POINT

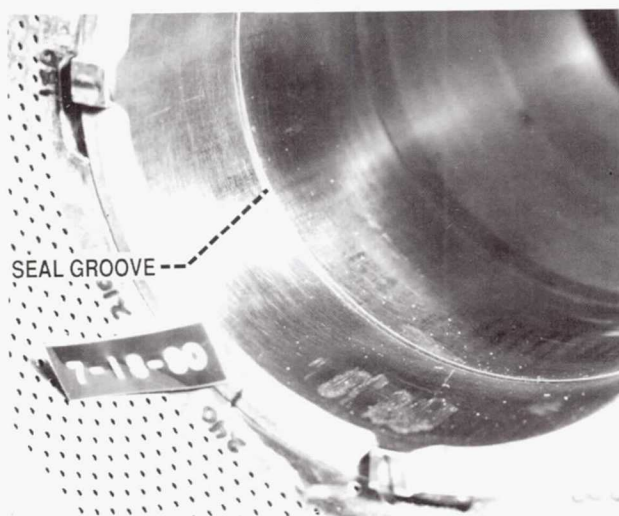| CENTERING PORTS	DISPLACER SEAL RING	PV POWER MEASURED (KW)	PV POWER PREDICTED (KW)	Δ PV POWER (KW)	PV EFF. MEASURED (%)	PV EFF. PREDICTED (%)	Δ PV EFF. (%)
6	NO	12.568	12.139	0.429	19.279	18.520	0.759
4	NO	12.395	12.066	0.329	19.476	18.506	0.970
6	YES	12.144	11.821	0.323	19.418	18.326	1.092
3	YES	11.649	11.376	0.273	18.759	17.915	0.844
2	YES	12.514	11.682	0.832	19.849	18.031	1.818

FIGURE 1. DISPLACER CYLINDER MODIFICATION.

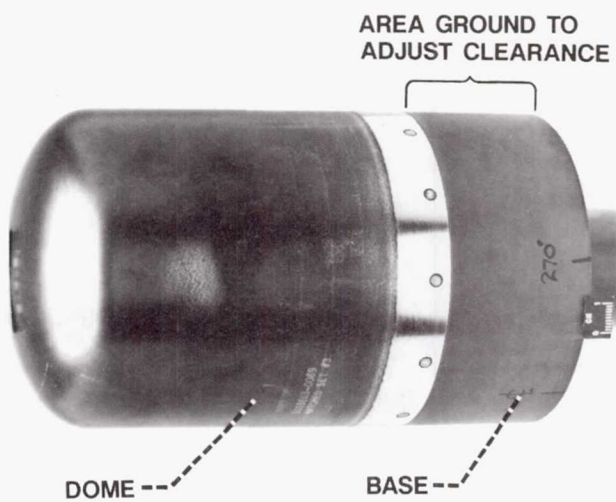
FIGURE 2. DISPLACER SEAL.

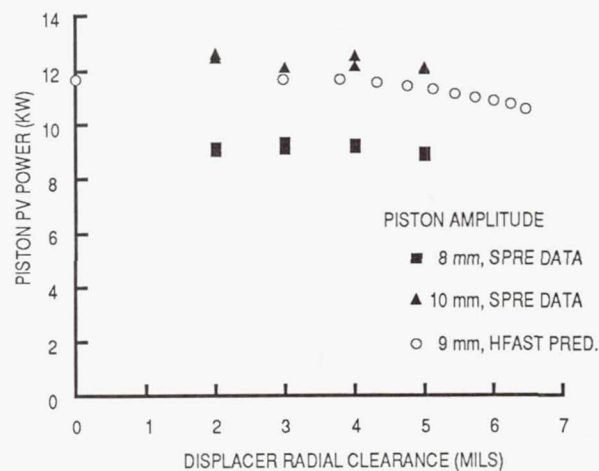FIGURE 3. SPRE DISPLACER.

FIGURE 4. EFFECT OF THE DISPLACER SEAL CLEARANCE ON PV POWER AT A TEMPERATURE RATIO OF 2.0 AND MEAN PRESSURE OF 15 MPa.
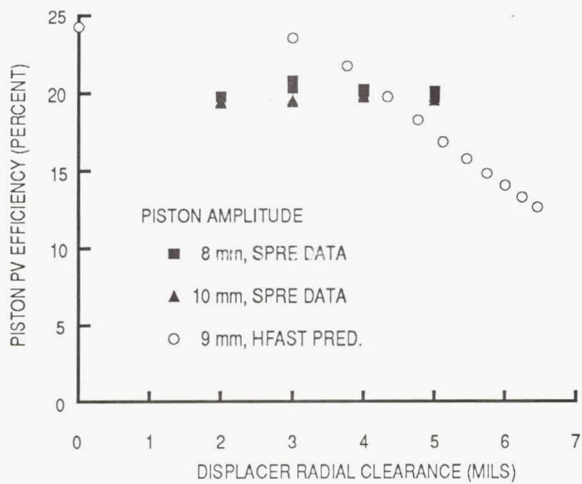
FIGURE 5. EFFECT OF THE DISPLACER SEAL CLEARANCE ON
PV EFFICIENCY AT A TEMPERATURE RATIO OF 2.0
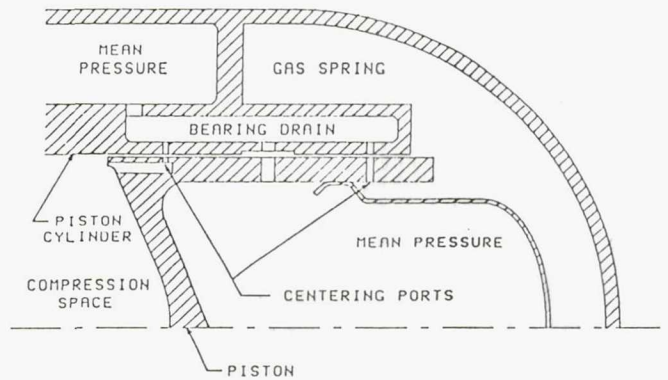AND MEAN PRESSURE OF 15 MPa.
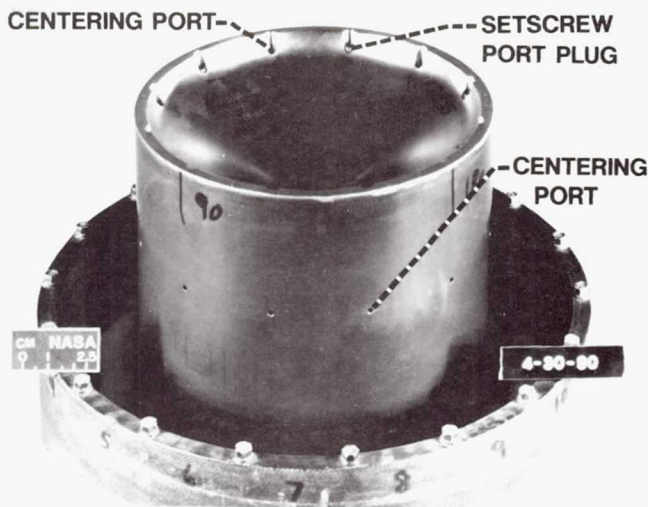

FIGURE 6. SPRE PISTON CENTERING PORTS.
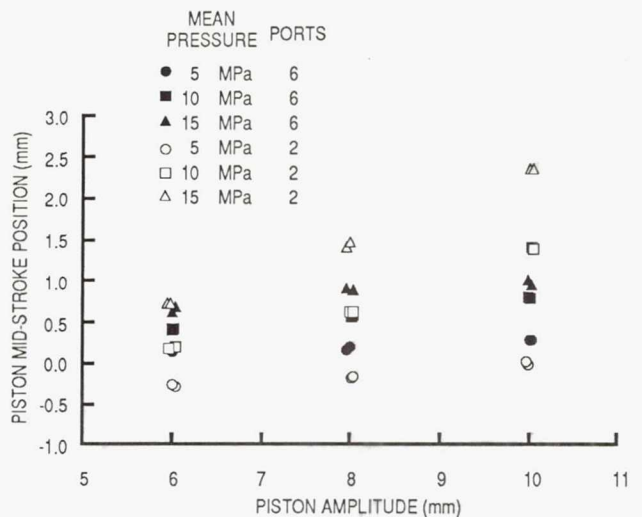

FIGURE 7. POWER PISTON ASSEMBLY

FIGURE 8. MEAN PRESSURE DEPENDENCE OF THE PISTON
MID-STROKE POSITION AT A TEMPERATURE RATIO OF 2.0.
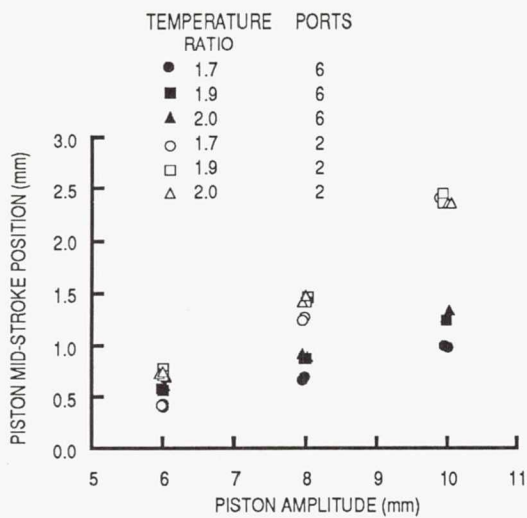
FIGURE 9. TEMPERATURE RATIO DEPENDENCE OF THE PISTON
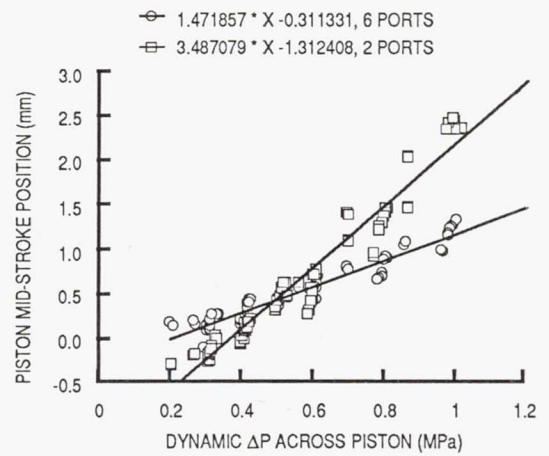MID-STROKE POSITION AT A MEAN PRESSURE OF 15 MPa.

FIGURE 10. PISTON MID-STROKE POSITION AS A FUNCTION
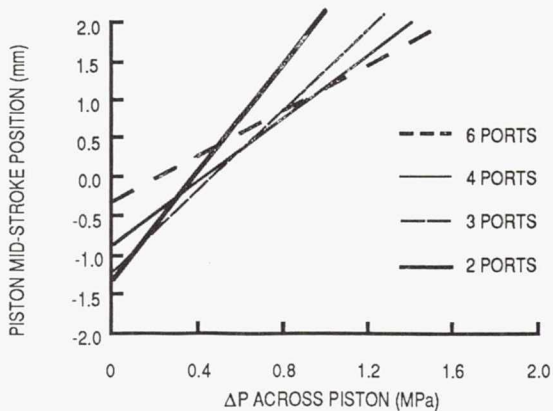OF THE DYNAMIC ΔP ACROSS THE PISTON

FIGURE 11. FIRST ORDER CURVE FITS OF THE DATA WITH
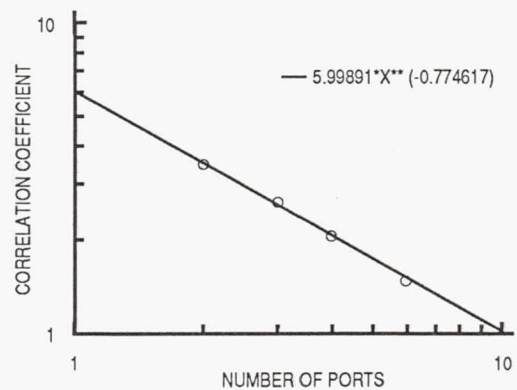6, 4, 3, AND 2 CENTERING PORTS OPEN.

FIGURE 12. DEPENDENCE OF THE CORRELATION COEFFICIENT
ON THE NUMBER OF OPEN CENTERING PORTS

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16. Abstract

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